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## SPEED EFFECTS ON BALL SPINNING TORQUE

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16. Abstract Tests were conducted in the NASA spinning friction apparatus to determine the effect of ball spinning speed on torque and coefficient of spinning friction. Reasonably good agreement was obtained between experimental torque values and theoretical torque predictions using an elastohydrodynamic analysis with a modified pressure-viscosity relation. Spinning torque increased with spinning speed, but the rate of increase was less than predicted from theory. This difference in the rate of increase in torque with speed was attributed to shearing of the lubricant which reduced the lubricant viscosity in the contact ellipse. No correlation was found between experimental and theoretical coefficients of spinning friction using a dry contact friction model.			
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## SPEED EFFECTS ON BALL SPINNING TORQUE

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### SUMMARY

The NASA spinning friction apparatus was used to determine the effect of ball spinning speed on torque and coefficient of spinning friction. SAE 52100 steel, 1/2-inch- (12.7-mm-) diameter balls were spun against race-groove specimens having ball-race conformities of infinity (ball on a flat), 60, 55, and 51 percent. The nominal hardness of the test specimens was Rockwell C 63. Test conditions were as follows: a maximum Hertz stress range of 50 000 to 300 000 psi ( $34.5 \times 10^7$  to  $207 \times 10^7$  N/m<sup>2</sup>), spinning speeds of 500 to 1500 rpm; room temperature (no heat added), and a synthetic paraffinic oil with no additives as the lubricant. Spinning torques were measured, and the coefficients of spinning friction were calculated.

Reasonably good agreement was obtained between experimental torque results and torque predictions using an elastohydrodynamic analysis with a modified pressure-viscosity relation. At maximum Hertz stresses of 80 000 to 150 000 psi ( $55.2 \times 10^7$  to  $103 \times 10^7$  N/m<sup>2</sup>) the rates of increase in spinning torque and coefficient of friction with spinning speed were less than predicted from theory. This deviation from the theory was attributed to shearing of the lubricant which reduced the lubricant viscosity in the contact ellipse.

### INTRODUCTION

Advanced turbine machinery for aircraft applications is expected to have larger diameter shafts and/or higher shaft speeds, which will require rolling-element bearings to operate at DN (bore diameter in mm times shaft speed in rpm) values of 2 to 4 million. One of the major problems anticipated for these high-speed rolling-element bearing applications is excessive heat generation which can cause large power loss and seizure of the bearing components (ref. 1).

Heat generated in a thrust-loaded ball bearing is caused by several factors which include rolling and spinning friction, viscous drag of the lubricant, and cage friction due to

sliding. The spinning of the ball in the Hertzian contact zone generates heat by the shearing of the lubricating fluid and from surface asperity contacts. The analysis now used to determine heat loss in ball bearings under thrust loads utilizes empirical equations similar to those that have been developed by A. Palmgren (ref. 2). In order to make a realistic analysis of bearing heat loss, the coefficient of friction of a ball spinning in a race, referred to as the coefficient of spinning friction, must be determined.

The coefficient of spinning friction also affects ball bearing kinematics. The current theory for predicting ball kinematics (refs. 3 and 4) assumes that a ball will roll without spin at one ball-race contact and have a combination of spinning and rolling at the other. This phenomenon is referred to as ball control. Rolling without spinning should theoretically occur at the ball-race contact at which the higher spinning moment develops. This moment is a function of the contact geometry and the coefficient of spinning friction.

The objectives of the research reported herein were to determine the effect of ball spinning speed on the spinning torque and coefficient of spinning friction with different stresses and ball-race conformities (the race-groove cross radius expressed as a percent of the ball diameter). Tests were conducted in the NASA spinning friction apparatus (ref. 5) at room temperature with ball-spinning speeds from 500 to 1500 rpm and stresses from 50 000 to 300 000 psi ( $34.5 \times 10^7$  to  $207 \times 10^7$  N/m<sup>2</sup>). Ball-race conformities of infinity (ball on a flat), 60, 55, and 51 percent were used. A synthetic paraffinic oil having no additives was used as the lubricant with SAE 52100 steel 1/2-inch- (12.7-mm-) diameter ball and groove specimens. Spinning-friction coefficients were calculated from measured spinning torques and stresses. All experimental results were obtained with lubricant from the same batch and specimens from the same heat of material.

## SYMBOLS

a	major semiaxis of contact ellipse, in. (m)
b	minor semiaxis of contact ellipse, in. (m)
b'	semiwidth of contact ellipse at y, in. (m)
E(k)	complete elliptic integral of second kind
F	friction force per unit width, lb/in. (N/m)
$f_s$	coefficient of spinning friction
h	film thickness, in. (m)
$h_1$	film thickness in contact ellipse, in. (m)
K	thermal conductivity, Btu/(sec)(ft)(°F) (J/(m)(sec)(K))
k	modulus of the elliptic integral E(k)

$M_s$	spinning torque, lb-in. (N-m)
$M_1, M_2$	friction torque, lb-in. (N-m)
$P$	normal load, lb (N)
$R_G$	radius of cylindrical groove, in. (m)
$r, \theta, y$	cylindrical coordinates
$S$	contact stress or pressure, psi ( $N/m^2$ )
$S_1$	pressure at which value of pressure-viscosity exponent changes, psi ( $N/m^2$ )
$v$	velocity, in./sec (m/sec)
$x, y, z$	Cartesian coordinates or distance, in. (m)
$\alpha, \beta$	pressure-viscosity exponent, $psi^{-1} (N/m^2)^{-1}$
$\gamma$	temperature-viscosity exponent, $^{\circ}F^{-1} (K)^{-1}$
$\mu$	absolute viscosity, reyns, lb-sec/in. <sup>2</sup> ( $N\text{-sec}/m^2$ )
$\bar{\mu}$	equivalent viscosity in the contact ellipse, reyns, lb-sec/in. <sup>2</sup> ( $N\text{-sec}/m^2$ )
$\mu_b$	equivalent viscosity at $y = b$ , reyns, lb-sec/in. <sup>2</sup> ( $N\text{-sec}/m^2$ )
$\mu_0$	absolute viscosity at atmospheric pressure and temperature, reyns, lb-sec/in. <sup>2</sup> ( $N\text{-sec}/m^2$ )
$\mu_x$	absolute viscosity at local pressure and temperature, reyns, lb-sec/in. <sup>2</sup> ( $N\text{-sec}/m^2$ )
$\psi$	function
$\omega$	angular velocity, rad/sec

## BACKGROUND

There are several factors that can affect ball spinning and rolling friction in an angular-contact ball bearing. Some of the parameters which have been reported to affect ball-spinning friction are ball-race conformity, contact stress, and lubricant type (refs. 5 to 7). Other factors that may affect the spinning friction are surface finish, spinning speed, lubricant viscosity, and other lubricant properties.

It was reported in references 6 and 7 that the coefficient of spinning friction decreases as the ball-race percent conformity decreases at a given stress condition. For a synthetic paraffinic oil, the minimum coefficient of spinning friction approaches a value of 0.05 at maximum Hertz stresses above 100 000 psi ( $69 \times 10^7 N/m^2$ ) for 51 percent and 300 000 psi ( $207 \times 10^7 N/m^2$ ) for 60 percent.

Combined rolling and sliding friction was measured between two disks in the work of reference 8. It was reported in reference 9 that calculated friction increases with sliding speed for combined rolling and sliding at low sliding speeds in the range of 0 to 20 inches per second (0 to 50 cm/sec) and decreases at higher sliding speed, in the range of 20 to 160 inches per second (50 to 400 cm/sec). It was also suggested in reference 9 that the friction traction per unit width can be described analytically by

$$F = \frac{2b\bar{\mu}v}{h_1} \quad (1)$$

The viscosity  $\bar{\mu}$  is dependent on several factors such as pressure, temperature, and shear rate.

The tests that were conducted in reference 8 show that the lubricant average film thickness does not change significantly with large changes in the equivalent viscosity in the contact zone. It was further reported that as the sliding speed increased from 0 to 158 inches per second (0 to 400 cm/sec) the equivalent viscosity in the contact ellipse decreased by a factor of 43 but the average film thickness decreased only 10 percent. In addition, the film thickness in the contact area was reported to be proportional to both rolling speed and inlet viscosity to the 1/2 power.

Ball-race friction is composed of rolling friction and spinning friction. Rolling friction is due largely to elastic deflection of the ball in contact with the race; spinning friction is due to (1) shearing of the lubricant under high-pressure, and (2) sliding under boundary lubricated conditions (ref. 10).

The combined rolling and spinning friction of balls rolling in a V-groove race (ref. 11) has been investigated. The force required to move a loaded upper V-groove race over balls free to rotate and in contact with a lower V-groove race was measured. For this case, the balls exhibited a combination of rolling and spinning. The spinning friction accounted for approximately 90 percent of the total friction. The results indicated that the spinning torque increased with decreasing ball-race percent conformity. The coefficient of spinning friction decreased with increasing maximum Hertz stress to 150 000 psi ( $103 \times 10^7$  N/m<sup>2</sup>). For further increases in stress the coefficient of friction tended to increase.

When equation (2) (ref. 10) and an assumed coefficient of friction are used, the torque due to spinning friction at a ball-race contact can be calculated:

$$M_s = 3/8 f_s PaE(k) \quad (2)$$

The friction coefficient  $f_s$  is assumed independent of the load and geometry for dry contact. For the lubricated case, this assumption is not valid since friction depends on the elastohydrodynamic conditions within the contact zone (ref. 12).

The work of reference 12 presents an elastohydrodynamic solution of torque for a ball spinning without rolling in a nonconforming groove. This work predicts spinning torques of the same order of magnitude using a modified pressure-viscosity exponential law as those obtained experimentally.

## APPARATUS AND PROCEDURE

A spinning friction apparatus (see fig. 1) as reported in references 6 and 7 was used for the tests reported herein. The apparatus essentially consists of a turbine drive, a pneumatic load device, an upper and lower test specimen, a lower test-housing assembly incorporating a hydrostatic air bearing, and a torque-measuring system. In operation, the upper test specimen is pneumatically loaded against the lower test specimen through the drive shaft. As the drive shaft is rotated, the upper test specimen spins in the groove of the lower test specimen. This causes an angular deflection of the lower test specimen housing. This angular movement is measured by the torque-measuring system and is translated into a torque value. During a test, the torque is continuously recorded on a strip chart.

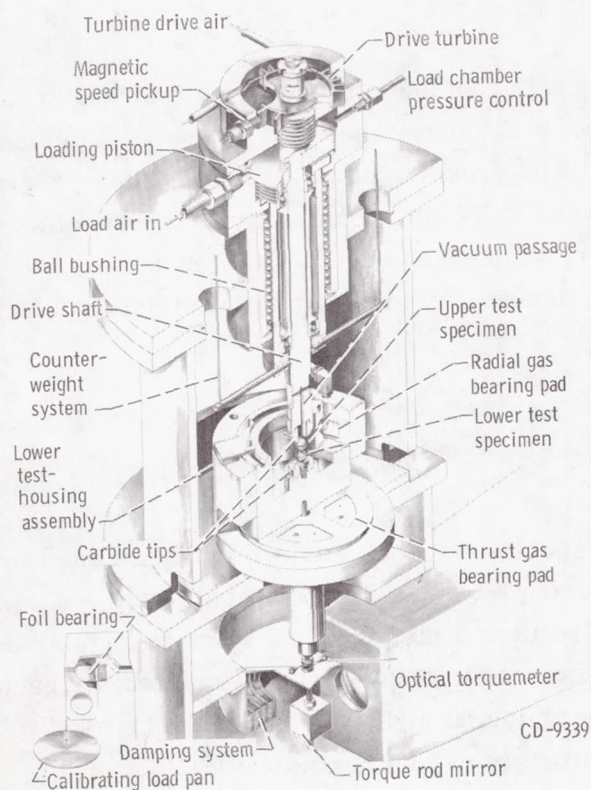


Figure 1. - Spinning friction apparatus.



## SPECIMENS

The upper test specimen is a conventional 1/2-inch- (12.7-mm-) diameter bearing ball made of SAE 52100 steel having a nominal Rockwell C hardness of 63 and a surface finish of 2 microinches ( $5\text{ }\mu\text{cm}$ ) rms. The lower test specimen (fig. 2) is a 1/2-inch- (12.7-mm-) diameter ball from the same heat of material as the upper test specimen and

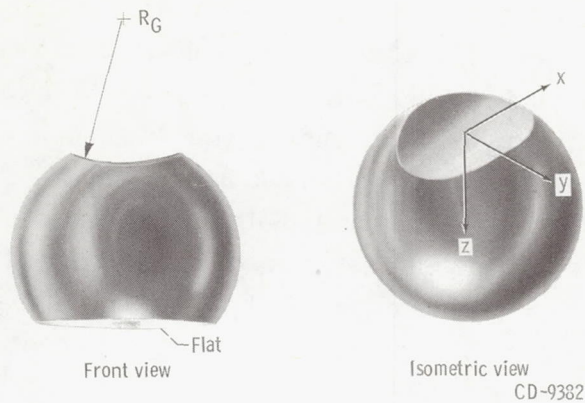


Figure 2. - Lower test specimen.

is modified by grinding a flat on one side and a cylindrical groove of radius  $R_G$  (fig. 2) on the other. The groove simulates the race groove of a bearing. The axis of the groove is parallel to the flat. The groove radius expressed as a percentage of the upper-ball diameter is defined as the ball-race conformity. The specimens used in these tests were ground to ball-race conformities of infinity (ball on a flat), 60, 55, and 51 percent. The surface finish of the cylindrical groove was approximately 2 to 6 microinches (5 to  $15\text{ }\mu\text{cm}$ ) rms.

## Operating Procedure

Prior to test, the specimens were ultrasonically cleaned in ethyl alcohol and vacuum dried for 24 hours. During a test, the experimental value of spinning torque was determined from a strip chart after a steady-state value of angular deflection was reached.

The spinning friction coefficient  $f_s$  was calculated for each test condition. A synthetic paraffinic oil containing no additives (table I) was used in all tests. All tests were run at ambient temperature (i. e., no heat added).



TABLE I. - VISCOSITY OF SYNTHETIC

PARAFFINIC OIL<sup>a</sup>

Temperature		Kinematic viscosity, cs
<sup>o</sup> F	K	
100	311	314
210	372	32

<sup>a</sup>No additives.

## RESULTS AND DISCUSSION

Tests were conducted with 1/2-inch- (12.7-mm-) diameter SAE 52100 steel balls spinning in grooved test specimens of varying conformity to determine the effect of speed on spinning torque and friction. Test conditions included spinning speeds from 500 to 1500 rpm and maximum Hertz stresses from 50 000 to 300 000 psi ( $34.5 \times 10^7$  to  $207 \times 10^7$  N/m<sup>2</sup>) with a synthetic paraffinic oil having no additive as the lubricant. The results were evaluated with respect to stress, ball-spinning speed, and ball-groove conformity. The resulting torques are presented in figure 3. These data correlate with those presented in references 6 and 7 showing increasing spinning torque with both increasing Hertz stress and decreasing percent conformity.

The coefficient of spinning friction was calculated for these torques using the equation from reference 10:

$$f_s = \frac{8/3 M_s}{PaE(k)} \quad (3)$$

These data are presented in figure 4. As was expected, the spinning friction values correlated with those of references 6 and 7. Converse to the torque values of figure 3, the coefficient of friction decreased with increasing Hertz stress and decreasing percent conformity.

## Speed Effects

In order to determine the effect of speed on spinning torque and coefficient of friction, values of torque from figure 3 and coefficient of friction from figure 4 were plotted as functions of spinning speed in figures 5 and 6 for several values of maximum Hertz stress. For the 51- and 55-percent conformities (figs. 5(a) and (b)) there appears to be

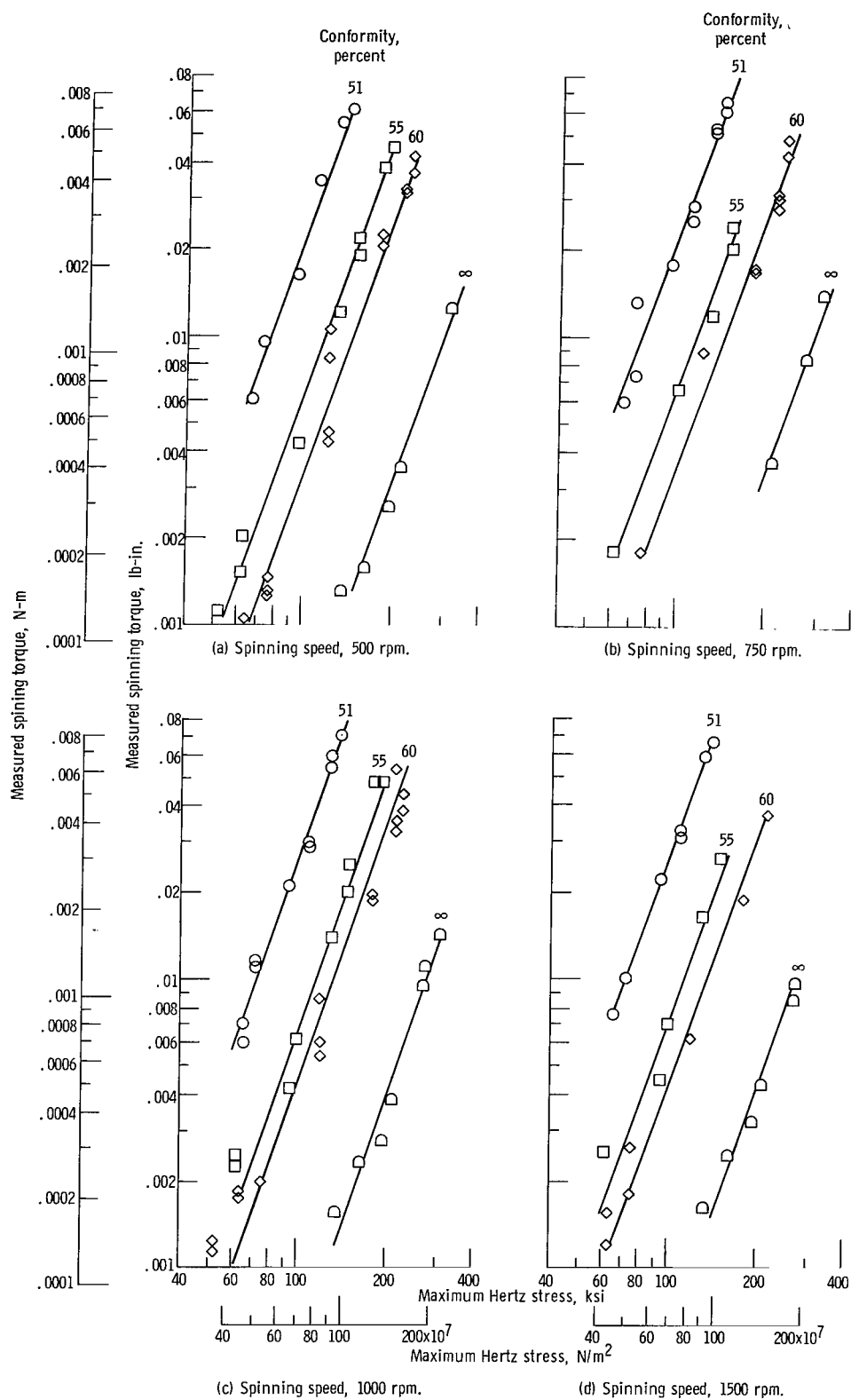


Figure 3. - Spinning torque as function of maximum Hertz stress. Lubricant, synthetic paraffinic oil; room temperature (no heat added).

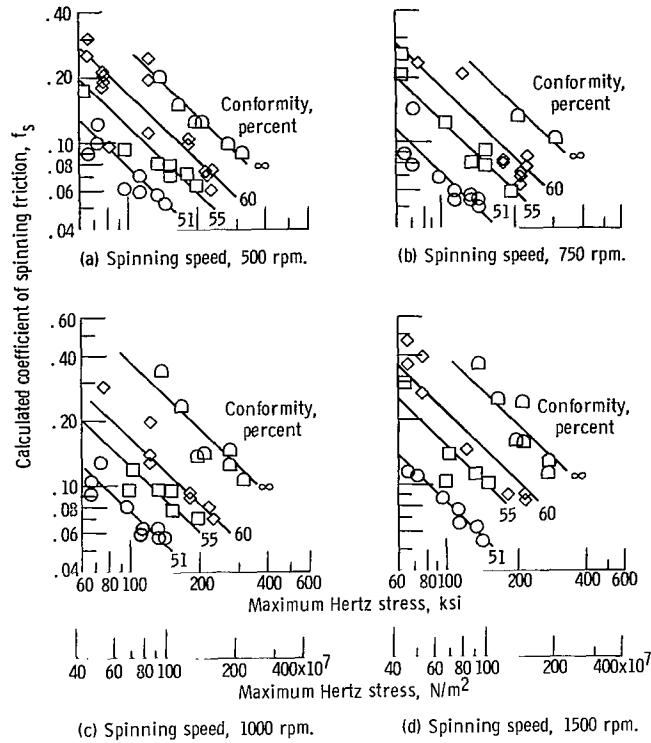


Figure 4. - Calculated coefficient of spinning friction as function of maximum Hertz stress. Lubricant, synthetic paraffinic oil; room temperature (no heat added).

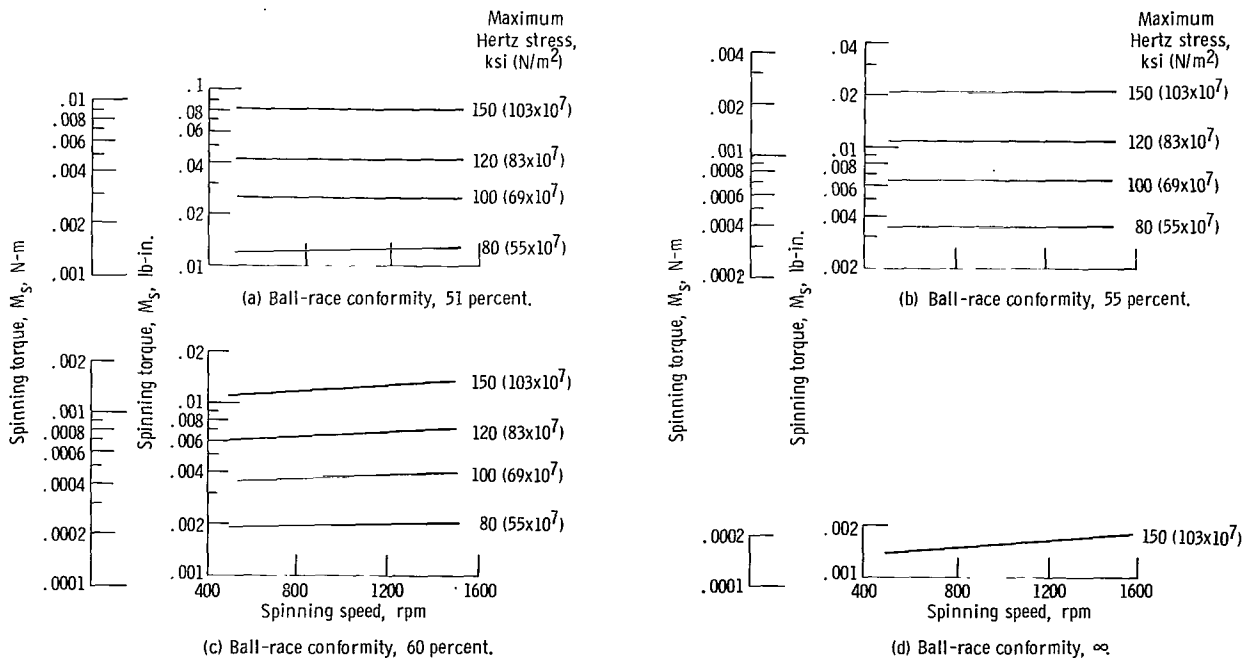


Figure 5. - Measured spinning torque as function of spinning speed. Lubricant, synthetic paraffinic oil; room temperature (no heat added).

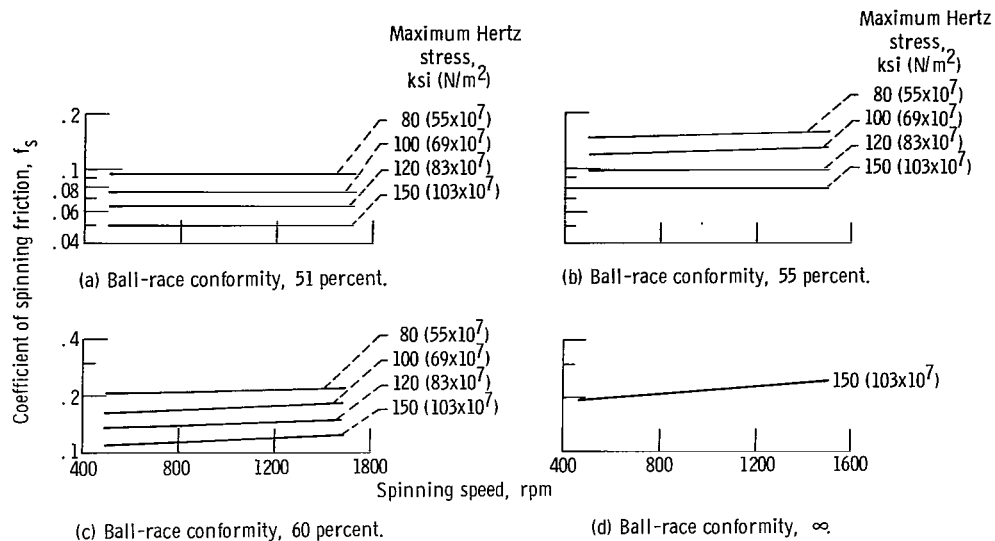


Figure 6. - Calculated coefficient of spinning friction as function of spinning speed. Lubricant, synthetic paraffinic oil; room temperature (no heat added).

no discernible effect of spinning speed on torque. At the 60-percent and infinite (fig. 5(c) and (d)) conformities there appears to be a very slight increase in torque with spinning speed.

The effect of speed on the coefficient of spinning friction (fig. 6) is somewhat more discernible. For the 51- and 55-percent conformity (fig. 6(a) and (b)) the coefficient of spinning friction appears to be constant. At 60-percent conformity (fig. 6(c)) a slight increase in friction with speed is shown and at infinite conformity (fig. 6(d)) (ball on a flat) the coefficient of friction increases with speed. The rate of increase with speed appears to be a function of conformity; friction increases at an increasing rate with speed as percent conformity increases.

## Theoretical Predictions

Current theory (ref. 10) assumes a constant coefficient of friction and unlubricated contact to determine ball spinning torque. The following analysis for spinning torque was developed (ref. 12) by using a theoretical model based on the elastohydrodynamic theory of lubrication.

Figure 7 is a representative contact ellipse for a ball in a nonconforming groove. The friction force  $M_1$  about the  $z$  axis due to the section of the ellipse for  $y > b$  and  $y < -b$  is obtained by integrating the product of the friction of the elemental roller (fig. 8)

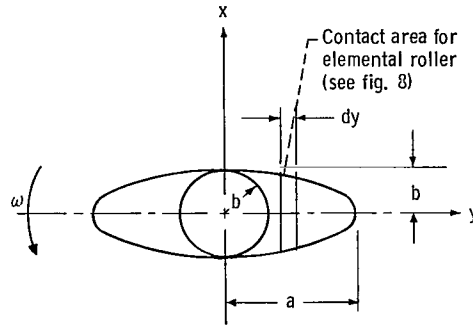


Figure 7. - Contact ellipse for ball in nonconforming groove (from ref. 12).

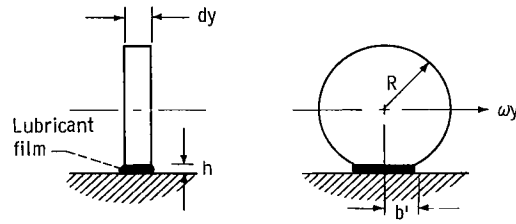


Figure 8. - Elemental roller (from ref. 12).

and the moment arm (ref. 12); thus,

$$M_1 = 2 \int_b^a y \, dF \quad (4)$$

where

$$dF = \frac{2b'\bar{\mu}\omega}{h} y \, dy \quad (5)$$

Because the form of the individual equations is such that a closed form solution does not exist, a numerical solution was adopted in reference 12 using Simpson's rule (ref. 13).

There remains the area within the inscribed circle where a true elastohydrodynamic film theory is impossible to maintain. Therefore, the film thickness  $h$  is assumed to be the same as that at  $y = b$ . If the inscribed circle is then divided into elemental rings of radius  $r$  and width  $dr$ , from reference 12, the torque  $M_2$  for the inscribed circle becomes

$$M_2 = \frac{2\pi\omega}{h} \int_0^b r^3 \mu_b \, dr \quad (6)$$

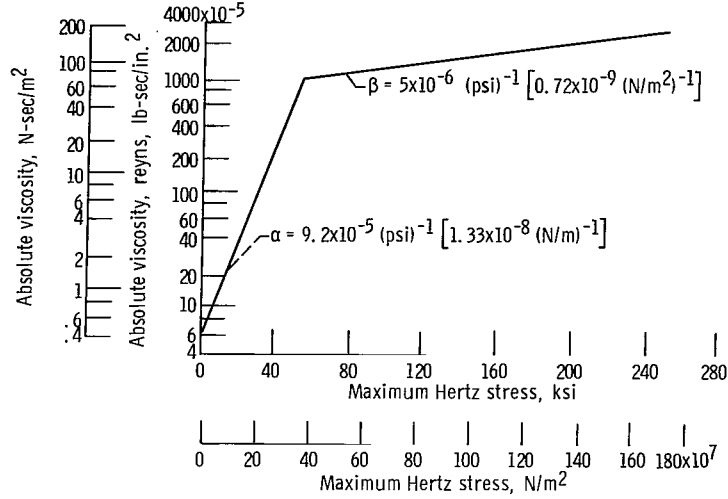


Figure 9. - Theoretical pressure viscosity relation for synthetic paraffinic oil at 83° F (302 K) (from ref. 12).

Summing equations (4) and (6) gives

$$M_S = M_1 + M_2 \quad (7)$$

Using a modified viscosity-pressure exponential law and the pressure-viscosity curve (ref. 12) of figure 9 give

$$\mu = \mu_o e^{\alpha S} \quad (8)$$

where  $S \leq S_1$ , and

$$\mu = \mu_o \exp [\alpha S_1 + \beta(S - S_1)] \quad (9)$$

where  $S > S_1$ ,  $\alpha = 9.2 \times 10^{-5} \text{ (psi)}^{-1} (1.33 \times 10^{-8} \text{ (N/m}^2\text{)}^{-1})$ ,  $\beta = 5 \times 10^{-6} \text{ (psi)}^{-1} (0.72 \times 10^{-9} \text{ (N/m}^2\text{)}^{-1})$ , and  $S_1 = 55 \text{ 000 psi } (38 \times 10^7 \text{ N/m}^2)$ .

The theoretical spinning torque as a function of speed for 51-, 55-, and 60-percent conformities was calculated. These values are shown in figure 10 for a maximum Hertz stress of 120 000 psi ( $83 \times 10^7 \text{ N/m}^2$ ). From these data there is a relatively good correlation between the experimental and theoretical values as a function of speed for the 55- and 60-percent conformities. A greater discrepancy exists between the theoretical and experimental values at the 51-percent conformity. At this conformity the correlation is better at 500 rpm than at 1500 rpm where the theoretical value is approximately twice that of the experimental value. Equation (1) indicates that torque and friction are proportional to speed and viscosity with the other parameters remaining nearly constant.



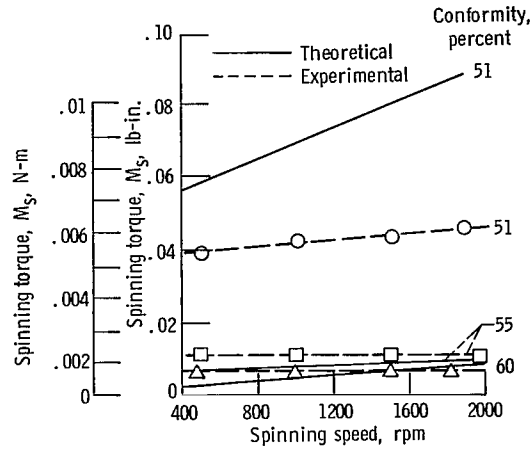


Figure 10. - Spinning torque as function of spinning speed. Lubricant, synthetic paraffinic oil; maximum Hertz stress, 120 000 psi ( $83 \times 10^7$  N/m<sup>2</sup>).

The experimental data suggest the effect of increasing speed is offset by the effect of decreasing viscosity. The decrease in viscosity is probably the result of shearing of the lubricant in the contact zone.

Sliding in the contact ellipse decreases the viscosity by viscous heating. For this case the local viscosity at  $y$  may be obtained by the following equation (ref. 12):

$$\mu = \frac{1}{b'} \int_0^{b'} \mu_x f(\psi) dx \quad (10)$$

where

$$\psi = \frac{\mu_x \gamma \omega^2 y^2}{8K} \quad (11)$$

and

$$f(\psi) = \frac{\ln(\sqrt{\psi + 1} + \sqrt{\psi})}{\sqrt{\psi(\psi + 1)}} \quad (12)$$

The spinning torque values were recalculated using effective viscosity equations (10) to (12) in the analysis. At a maximum Hertz stress of 150 000 psi ( $103 \times 10^7$  N/m<sup>2</sup>), a 51-percent conformity, and a speed of 500 rpm, viscous heating had an insignificant effect on the value of torque calculated without viscous heating effect. Under the same

conditions, but at a speed of 1500 rpm, the viscous heating reduced the calculated torque by only 3 percent. It is apparent that viscous heating, as defined by equations (10) to (12), does not account for the difference between the experimental and theoretical torque results.

There are several possible explanations for this deviation from the theoretical predictions. It is possible that equations (10) to (12) are not applicable to the case of a ball spinning in a groove without rolling. In addition, these equations may not account for all the heat generated within the contact zone. There also exists the probability that the synthetic paraffinic oil, which is the lubricating fluid, acts in a non-Newtonian manner at the higher spinning velocity. Either one or all of these may be valid explanations. However, until the exact rheological behavior of the fluid is defined, less speculative answers cannot be furnished.

When both the experimental and theoretical values of spinning torque were used, the coefficients of spinning friction were calculated using equation (2) for 51-, 55-, and 60-percent conformities, a maximum Hertz stress of 120 000 psi ( $83 \times 10^7 \text{ N/m}^2$ ), and spinning speeds of 500 to 1500 rpm. These values of spinning friction are shown in figure 11.

A large discrepancy exists between the friction values calculated from the theoretical and experimental torques even though there is a reasonably good correlation between the torque values. For the experimental results the coefficient of spinning friction decreases with decreasing percent conformity. The opposite occurs for the theoretical results.

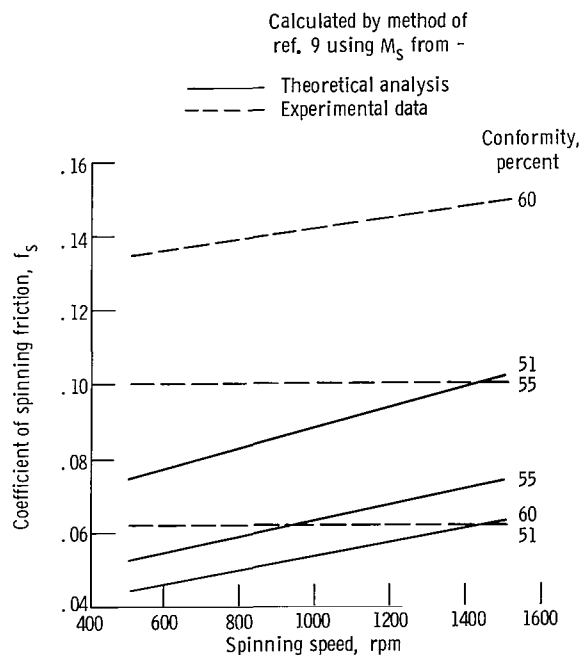


Figure 11. - Spinning friction as function of spinning speed. Lubricant, synthetic paraffinic oil; maximum Hertz stress, 120 000 psi ( $83 \times 10^7 \text{ N/m}^2$ ).

From equation (2),  $f_s \propto [M_s/PaE(k)]$ . For the experimental results, the value for  $[PaE(k)]$  increases at a greater rate than torque  $M_s$  as percent conformity decreases. Hence, the lower the percent conformity the lower the coefficient of spinning friction. The theoretical analysis predicts a greater rate of increase in torque  $M_s$  than the value  $[PaE(k)]$  with decreasing percent conformity. Thus, for the analytical results the lower conformity has the highest coefficients of friction.

From the data contained herein and in references 6 and 7, a constant coefficient of spinning friction cannot be assumed under lubricated contacts. In addition, the use of a variable coefficient of friction based on equation (2) and the theoretical analysis for spinning torque will not yield results which conform with experimental data. This would suggest that in the determination of ball bearing kinematics an elastohydrodynamic (EHD) solution must be incorporated in place of equation (2). In addition, the concept of using a coefficient of spinning friction should not be incorporated in any EHD solution.

## SUMMARY OF RESULTS

The NASA spinning friction tester was used to determine the effect of spinning speed on spinning torque. SAE 52100 steel 1/2-inch- (12.7-mm-) diameter balls were spun against race-groove specimens having ball-race conformities of infinity (ball on a flat), 60, 55, and 51 percent. The nominal hardness of the test specimens was Rockwell C 63. Test conditions were as follows: a maximum Hertz stress range of 50 000 to 300 000 psi ( $34.5 \times 10^7$  to  $207 \times 10^7$  N/m<sup>2</sup>), spinning speeds from 500 to 1500 rpm, room temperature (no heat added), and a synthetic paraffinic oil with no additives as the lubricant. Spinning torques were measured and the coefficients of spinning friction were calculated. The following results were obtained:

1. The rate of increase in experimental spinning torque with spinning speed was less than expected from theory. This was attributed to the shearing of the lubricant which reduced lubricant viscosity. The lower the percent conformity, the higher the rate of torque increase.

2. When the elastohydrodynamic theory and a modified pressure-viscosity relation were used, reasonably good agreement was found between theoretical and experimental values of spinning torque.

3. A large discrepancy existed between the friction values calculated from a dry friction model using the theoretical and experimental torques. The coefficient of spinning friction calculated from the experimental torque values decreased with decreasing percent conformity. The opposite occurs for friction values calculated from theoretical

torque results. This result suggests that the concept of using an equation for a coefficient of spinning friction based on unlubricated surfaces in contact cannot be used under elastohydrodynamic conditions.

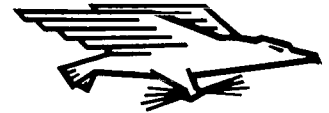
Lewis Research Center,  
National Aeronautics and Space Administration,  
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